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EFFECT OF CHAMFERED HOLES ON THE RESISTANCE OF BOLTS AND DOWELS TO SHOCK LOADS IN SHEAR

I. Vigness, E. R. Seibert, and H. M. Forkois

Shock and Vibration Branch
Mechanics Division

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U. S. NAVAL RESEARCH LABORATORY
Washington, D.C.

CONTENTS

ABSTRACT

Under dynamic loading conditions associated with shock motions, hold-down bolts may be subjected to both shear and tensile stresses. If the bolt holes in the shear plane are chamfered so as to permit the bolt to deform in shear and bending along its length, the amount of energy absorbed by the bolt before fracture is increased when shear motions are encountered. The amount of energy absorbed when a 45-degree chamfer was employed was found to be a constant (the energy absorbed by a bolt in shear with no chamfer) plus a term directly proportional to the length of the chamfer. The maximum load developed is about the same for all chamfer sizes in a given material, but this load will be maintained for a greater number of shocks for the larger chamfer sizes. The chamfer caused some decrease in stiffness of the joint for shear motions. For relatively large chamfer sizes, or for relatively non-ductile materials, bolt failure resulting from shear motions may be prematurely caused by tensile stresses.

PROBLEM STATUS

This is a final report on this phase of the problem; work on other phases continues.

AUTHORIZATION

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INTRODUCTION

Hold-down bolts are sometimes required to maintain the position of an equipment so that it will remain in alignment with other apparatus. These bolts are always required to secure the equipment so that it will not become loose and collide with other objects or cause injury to personnel. Under conditions of mechanical shock it is possible to design equipment hold-down bolt connections so that plastic shear deformations occur in the bolts, thereby reducing the probability of bolt shear fractures resulting in complete failure. For a given number and size of bolts and arrangement of bolt holes, this can be accomplished by chamfering the edges of the bolt holes on the plane surfaces of the members held together by the bolts. These chamfers increase the shock-to-fracture resistance of the bolts since they allow large plastic shear deformations before shear fracture begins, and therefore the plastic range to ultimate fracture is extended. The use of chamfered bolt holes for greater shock-to-fracture resistance of bolted connections results in slightly less stiffness or resistance to misalignment for relatively small loads, but there is little difference between chamfered and nonchamfered holes in the maximum static load that can be supported. The plastic shear motions that can take place with the chamfer require forces in the same range as those which would ordinarily cause failure without the chamfer. As the shear deformation permitted is much greater in the former case, the bolt will absorb more energy before fracture. Where alignment is critical it may be advisable to use dowels in a conventional manner and chamfer only the bolt holes.

EXPERIMENTAL WORK

Two types of experiments were performed. In the first, which will be called the "bolt test," the ability of bolts to withstand loads imposed by the Navy High-Impact Shock Machine for Mediumweight Equipment was determined. A bolt held two plane plates together. The plates were constrained so as to permit motion only in a direction parallel to their contacting surfaces. One plate was attached to a mass which provided an inertial load to the bolt and the other plate was attached to the shock machine mounting channels.

In the second experiment, which will be called the "dowel test," dowel-like specimens were tested in shear to determine their ability to absorb the impact energy of a mass that was dropped a given distance.

The principal difference between the two tests was in the bolt head and nut, which caused tensile stresses in the bolt which were not present in the dowel. The experimental conditions and results are presented separately for each of these conditions.

BOLT TESTS

The bolt tests described here are essentially the same as those described in a preliminary report (1) to the Bureau of Ships. At the request of the Bureau (2), shock tests were conducted according to a modification of military specification MIL-S-901B(Navy) in a manner that caused the bolt to be loaded principally in shear.

The experimental arrangement is shown in Fig. 1. The bolt specimens were inserted through two steel bushings which were placed in two 8 × 6-1/16 × 1-7/8 in. steel plates; the junction of these bushings formed the shear plane. The friction-bearing area of this plane was approximately 35 square inches. Three-quarter- and 1-3/4-in.-diam Navy Grade-2 (Military Specifications MIL-B-857 Bolts, Nuts, and Studs) bolts were used. The yields and tensile strengths of the bolt material were 52,000 psi yield and 64,000 psi tensile for the 3/4-in.-diam bolts, and 28,000 psi yield and 55,000 psi tensile for the 1-3/4-in.-diam bolts. The bushings, which contained the chamfered surfaces, were made of SAE 4340 steel and treated to a hardness of about 35 Rockwell. Four support brackets, fastened to the ends of the mounting channels, supported a guide frame. This frame guided the upper hold-down plate, to which was fastened the upper shear-plate, so as to permit vertical and prevent lateral motion, thus allowing the shearing action to take place in the bolt. Weights were bolted to the upper hold-down plate to provide the required load on the bolt. The lower shear-plate was rigidly fixed to the lower hold-down plate, which in turn was bolted to the mounting channels of the shock machine for mediumweight equipment.

One bushing was placed in the upper shear-plate and the other in the lower shear-plate (Fig. 1). The bolt specimen was then pushed through the bushings and the nut tightened to the proper value of torque. All bushing and bolt fits were class 3. An initial micrometer measurement was taken to determine the distance between the upper hold-down plate and the top of the lower shear plate. The specimen was then subjected to the required hammer drop and the two velocity curves were recorded. The overall shear displacement of the bolt was then determined from the micrometer measurements. In addition, the torque required to tighten the bolt to an acceptable value was measured. This procedure was followed until the bolt fractured.

Several different combinations of bushings with different chamfers were used. These combinations were as follows: no chamfer on either bushing; one side with no chamfer, the other side with a 1/16-in., 45-degree chamfer; two sides with a 1/16-in., 45-degree chamfer; one side with no chamfer, the other side with a 1/8-in., 45-degree chamfer; two sides with a 1/8-in., 45-degree chamfer.

The shock tests for seven specimens (specimens I to VI and XII of Table 1) were administered in accordance with specifications MIL-S-901B. The remaining five specimens (VII to XI) were subjected to successive 1.0-ft blows until fracture occurred. A predetermined weight was placed on each bolt. For specimens I, II, and III this load was 654 lb, in accordance with the static loading of a particular shipboard installation. For specimens IV to XII this load was reduced to 432 lb.

Two velocity pickups were mounted on the test apparatus, as shown on Fig. 1. One, and MB type 200, was mounted on top of the weights which were bolted to the upper hold-down plate. The second, a Hartz type, was mounted directly to the lower hold-down plate. Simultaneous recordings of the signals produced by these pickups were made on a Visicorder for each hammer drop. A typical tracing of a short section of each of these is shown in Fig. 2. The data obtained from these records are summarized in Table 1. The force produced on the bolt by the shock load can be found by multiplying the total weight supported by the bolt by the initial slope (acceleration) of the load velocity curve, this being expressed in units of gravity, g. The slope taken is an average value for the time to maximum velocity and is shown by the dotted lines on Fig. 2. Examples of fractured 3/4-in. bolts are shown on Fig. 3. The heads of the bolts had to be cut off in order to remove the bolt from the bushing. The test number, bushing specification, number of blows, hammer height for each blow, table travel, acceleration, force, and deflection are tabulated in Table 1.

The 3/4-in. bolt specimens did not withstand the shock specifications (MIL-S-901B) for the weight of 654 lb under any condition or combinations of chamfering. After the test

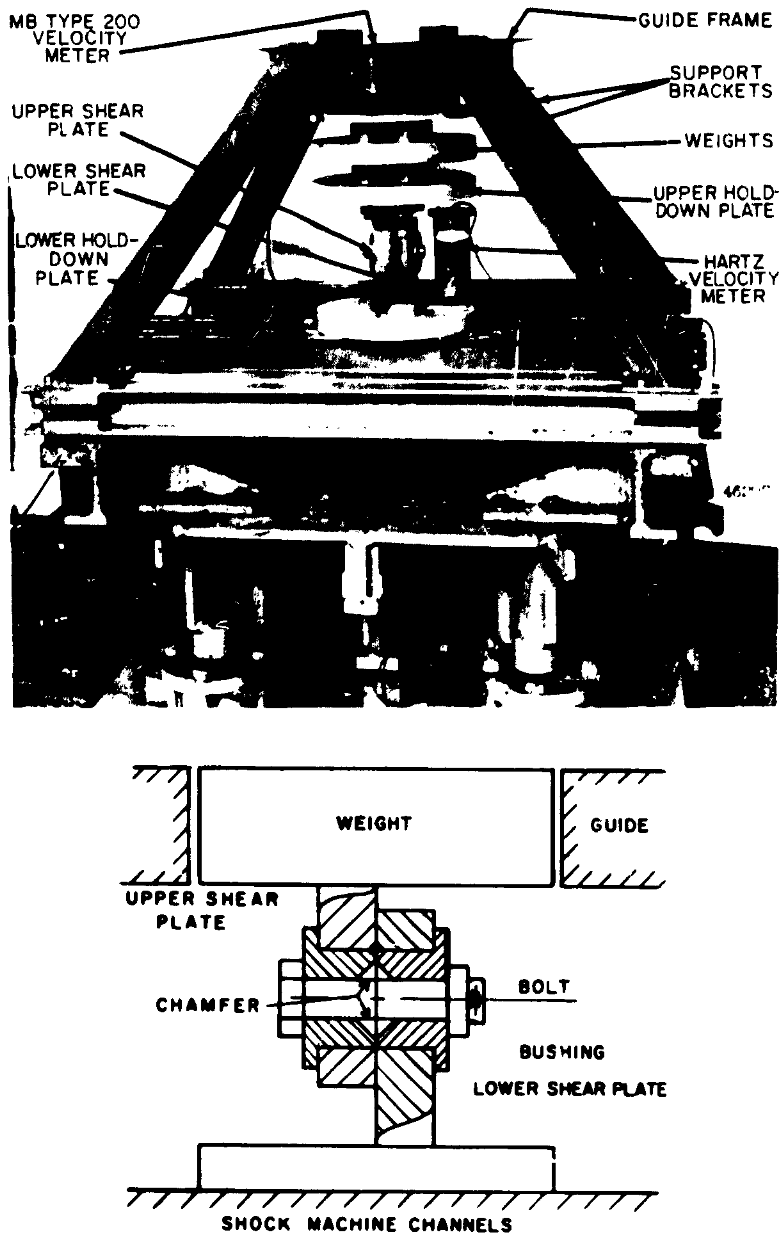


Fig. 1 - Shear test fixture secured to mounting channels attached to the anvil table of the Navy High-Impact Shock Machine for Mediumweight Equipment. The insert shows details of the arrangement.

of the third specimen it was concluded that the load of 654 lb was too large for experimental purposes. For this reason the weight was reduced to 432 lb. Even with this load, bolt failure occurred before the completion of the specification test. Bolt specimens VII to XI were therefore submitted to successive 1.0-ft hammer drops in order to obtain information regarding the effects of the chamfered bushings. Each blow resulted in plastic deformation of the bolt. As a result, the nut on the bolt was loose and was tightened to a torque of 140 lb-ft after each test.

Table 1
Results of Shear Tests on 3/4-Inch Grade-2 Bolts

G_M, G_B = accelerations, in units of gravity, of the weight supported by the bolt and the excitation given the bolt; they are average values based on the rise-time to maximum velocity.
F = the effective dynamic load in pounds supported by the bolt and is equal to G_M times the weight supported by the bolt.
ΔX_S = the shear displacement (in inches) resulting from a given blow.
$\Delta X_{cuml.}$ = the total cumulative shear displacement in inches.

Test Number	Blow Number	Hammer Height (ft)	Table Travel (in.)	G_M (g)	G_B (g)	F (lb)	ΔX_S (in.)	$\Delta X_{cuml.}$ (in.)	Remarks
I	1*	1.25	3	35	43	22,900	—	—	No chamfer
II	1	1.25	3	32	35	20,950	0.14	0.14	1/16″-45° chamfer, one side
III	1	1.25	3	34	42	22,210	0.15	0.15	1/16″-45° chamfer, both sides
IV	1	1.0	3	37	36	16,000	0.053	0.053	No chamfer
	2	1.0	3	54	40	23,300	0.054	0.107	
	3*	2.0	3	37	60	16,000	—	—	
V	1	1.0	3	38	33	16,410	0.077	0.077	1/16″-45° chamfer, one side
	2	1.0	3	47	33	20,300	0.053	0.130	
	3*	2.0	3	45	59	19,430	—	—	
VI	1	1.0	3	40	20	17,270	0.071	0.071	1/16″-45° chamfer, both sides
	2	1.0	3	55	11	23,760	0.037	0.108	
	3*	2.0	3	45	24	19,430	—	—	
VII	1	1.0	3	36	22	15,550	0.086	0.086	No chamfer
	2	1.0	3	58	32	25,040	0.048	0.134	
	3*	1.0	3	23	23	9,930	—	—	
VIII	1	1.0	3	54	27	23,300	0.087	0.087	1/16″-45° chamfer, one side
	2	1.0	3	55	28	23,750	0.036	0.123	
	3	1.0	3	66	34	28,500	0.014	0.137	
	4	1.0	3	57	35	24,610	0.052	0.189	
	5*	1.0	3	31	32	13,380	—	—	
IX	1	1.0	3	44	22	19,000	0.088	0.088	1/16″-45° chamfer, both sides
	2	1.0	3	58	29	25,050	0.032	0.120	
	3	1.0	3	73	58	31,540	0.010	0.130	
	4	1.0	3	76	39	32,800	0.056	0.186	
	5	1.0	3	74	29	31,950	0.029	0.215	
	6*	1.0	3	55	30	23,750	—	—	
X	1	1.0	3	42	33	18,150	0.090	0.090	1/8″-45° chamfer, one side
	2	1.0	3	62	39	26,790	0.042	0.132	
	3	1.0	3	54	28	23,310	0.083	0.215	
	4*	1.0	3	55	35	23,740	—	—	
XI	1	1.0	3	44	28	19,020	0.100	0.100	1/8″-45° chamfer, both sides
	2	1.0	3	49	31	21,170	0.042	0.142	
	3	1.0	3	62	36	26,790	0.057	0.199	
	4	1.0	3	63	32	27,200	0.014	0.213	
	5	1.0	3	69	41	29,810	0.051	0.264	
	6	1.0	3	59	39	25,490	0.022	0.286	
	7	1.0	3	85	32	36,720	0.048	0.334	
	8	1.0	3	72	35	31,100	0.041	0.375	
	9*	1.0	3	76	44	32,800	—	—	
XII	1	1.0	3	46	40	19,860	0.100	0.100	1/8″-45° chamfer, both sides
	2	1.0	3	56	38	24,190	0.041	0.141	
	3	2.0	3	62	42	26,800	0.091	0.232	
	4*	2.0	3	62	52	26,800	—	—	

*Fractured.

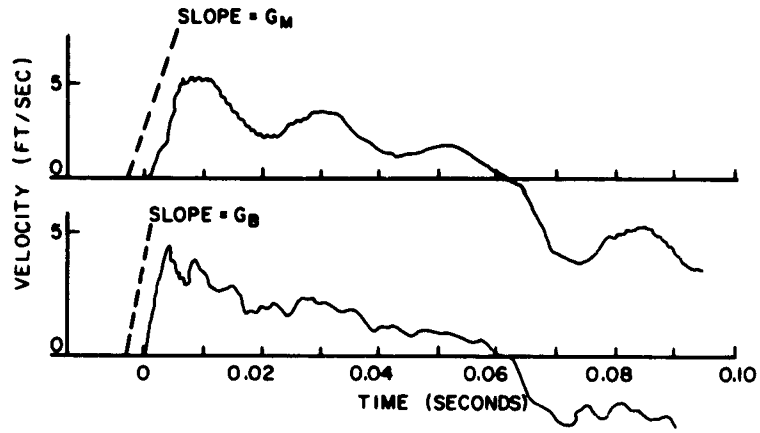


Fig. 2 - Example of velocity-time curves for (a) mass supported by bolt, and (b) excitation provided the bolt by shock machine

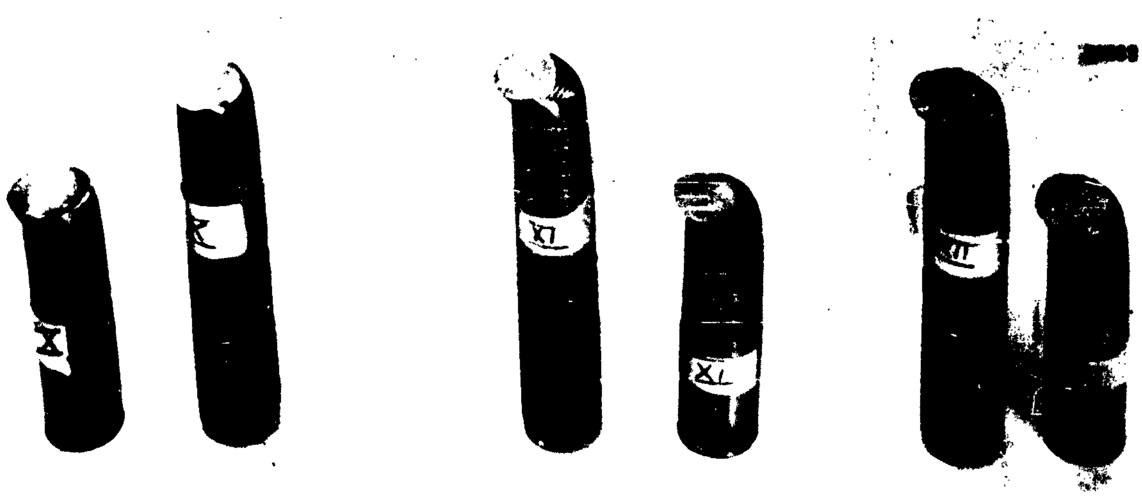


Fig. 3 - Typical examples of sheared bolts after removal from bushings

Using data in Table 1 the curves in Fig. 4 were constructed. These curves show the number of 1-ft hammer drops survived by bolts for various chamfers. It is apparent that chamfering is effective in increasing the fracture resistance of the bolt for these conditions of shock. The relationship between the chamfer size and the energy absorbed by a bolt is shown in Fig. 5. The energy values were each obtained as the summation for a series of n blows of the product of the inertial mass, the acceleration, and the corresponding increment of permanent deformation of the bolt for each blow. Since the inertial mass was 432 lb for each blow, a particular energy value is equal to

$$432 \sum_n G_{M_n} \Delta X_{S_n},$$

where G_{M_n} is the average value of acceleration of the mass (see Fig. 2) in units of gravity for the nth hammer blow, and ΔX_{S_n} is the shear deformation caused by the nth hammer blow.

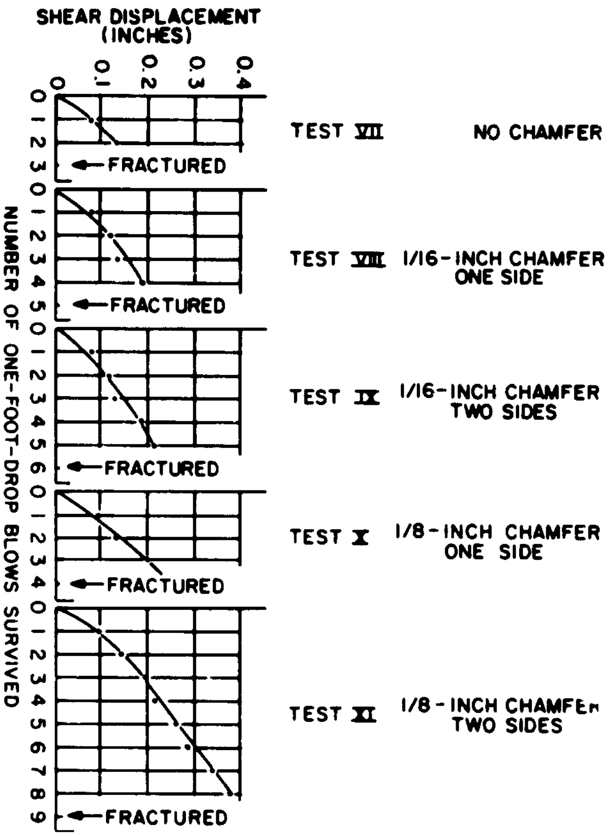


Fig. 4 - The relationship between the cumulative overall shear deflection of a bolt and the number of 1-ft hammer blows survived, for chamfer conditions as indicated

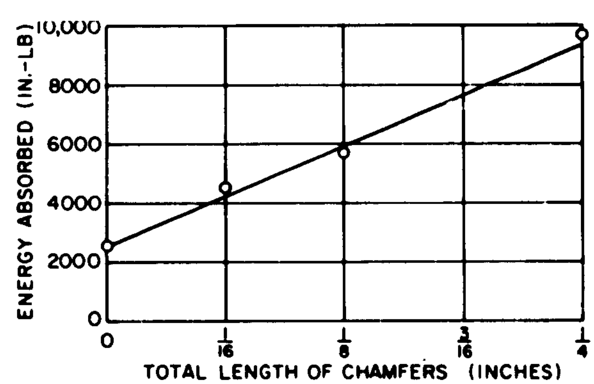


Fig. 5 - The relationship between the energy absorbed by 3/4-in.-diam bolts and chamfer length

Considerable experimental difficulties were encountered with the tests of the 1-3/4-in. bolts. These difficulties were the result of the considerable strength and stiffness of the bolts as compared with those of the fixtures. These results do not, therefore, have the quantitative accuracy of those for the 3/4-in. bolts.

The 1-3/4-in.-diam bolts supported a static load of 1550 lb. The nut, which loosened as the result of each test, was tightened to a torque of 1100 lb-ft after each test. The bolt with this load successfully passed the standard specification tests.

Because of the separation and change of angle between the shear surfaces of the upper and lower shear plate (Fig. 1) when the relatively large 1-3/4-in. bolts were used, the effect of the chamfer was not significant. The number of blows to failure, the amount

of energy absorbed, and the amount of shear displacement did not vary in a significant way with the amount of chamfer. When this bolt-diam requirement might arise during a field installation then the chamfer would be of negligible value.

DOWEL TESTS

A drawing of the apparatus used for subjecting the dowel specimens to shear loads is given in Fig. 6. It consists of a hammer that can be raised to any desired height and dropped upon an anvil. The two legs of the hammer straddle the yoke arrangement so as to strike the mating anvil surfaces. The anvil and yoke arrangement is supported by the dowel in double shear. Four sections of No. 18 B & S gage copper wire, totalling 18-3/4 inches in length, were placed on the anvil striking surfaces. These provided a smooth rate-of-increase of load under impact, thereby decreasing the excitation of vibratory transients.

The test fixture and dowel are shown disassembled in Fig. 7. The copper shims shown in that figure were used to provide for the proper fit between the yoke, yoke caps, and yoke bushings. Bushings were provided for the two classes of fits, commercial "close" and tight. An F-drill (diameter = 0.2570 in.) was used for the commercial "close" fit and provided a diametral clearance of about 0.007 in., and a 1/4-in. reamer (diameter = 0.2503 in.) was used for the tight fit and provided a diametral clearance of about 0.001 in. Four pairs of yoke bushings were provided for each of the two classes of fits, a pair for each of four chamfers.

Measurements of load versus time were made by means of a load sensor. The yoke section of the test fixture was screwed to the load sensor, which in turn was attached to the anvil. The load sensor consisted of four strain gages connected in a Wheatstone bridge. A 1-kc signal timing trace was also provided on the records. The residual plastic deformation was measured after each blow by measuring the change in separation of two gage dowels (Fig. 7), one attached to the yoke and the other to the rod.

Specimens were prepared from SAE 1030 hot-rolled round stock, 3-1/4 in. in diameter. This was forged to provide 9/16-in.-diam blanks. These blanks were then normalized at 1700°F to produce a homogeneous structure and to eliminate grain orientation. They were then quenched in water from 1600°F and drawn at 1000°F. The blanks were then machined, ground, and polished to the proper diameter, 0.2448 and 0.2491 in. A reduced size tensile specimen of 1-in. gage length prepared from one of the blanks had a yield strength of 76,300 psi for a 0.2 percent offset.

Additional specimens were cut from 1/4-in.-diam SAE 1020 cold-rolled rod, as received. These were lightly polished before use so that their diameter was 0.2493 in.

Static tests were made for all combinations of the two materials, two diametral clearances, and four chamfers. The same test fixture (Fig. 7) used for the dynamic tests was used for "static" tests in a Baldwin Southwark Tate-Emery Testing machine. As an illustration of the static shear tests, the load vs deflection curves for the SAE 1030 dowel specimens with diametral clearances of 0.0072 in. are presented in Fig. 8. Four curves are shown, one for each of the four chamfer lengths. From an examination of these curves, it can be seen that the specimens with the smaller chamfers developed their maximum loads sooner and fractured at smaller deflections. The stiffness of the joint for small (elastic) deflections generally decreases as the chamfer size increases. However, the lower bolt shear stresses resulting from the use of a small chamfer may more than compensate for the slightly greater deflections it may cause. The effect of chamfer on the maximum load is not appreciable; all specimens of the same material developed about the same maximum load. The bolts and dowels were all made of relatively ductile materials.

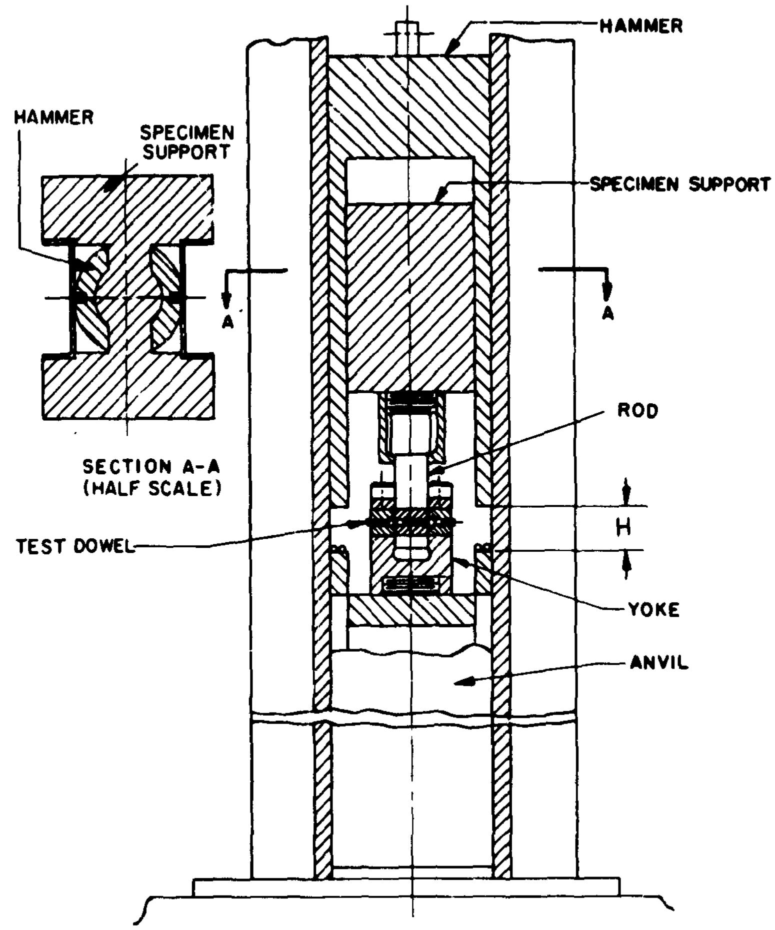


Fig. 6 - The experimental arrangement for testing dowels in shear for impact loads

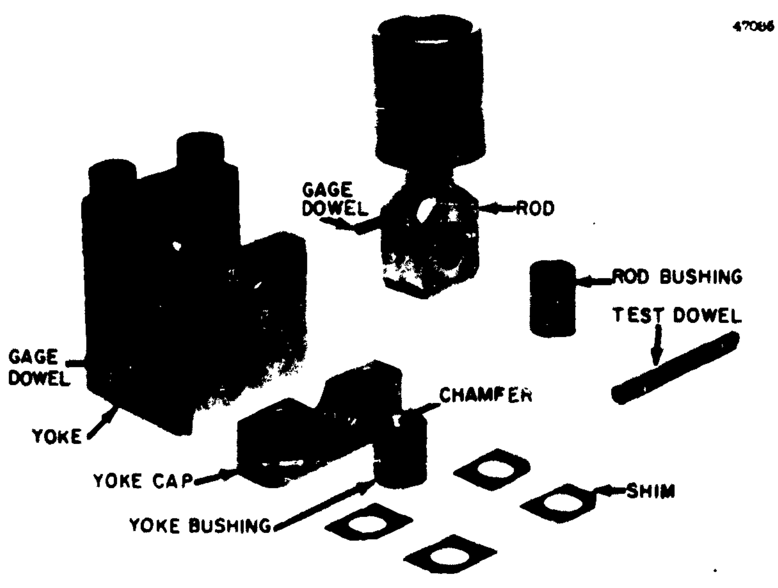


Fig. 7 - Disassembled fixture for testing dowels in shear

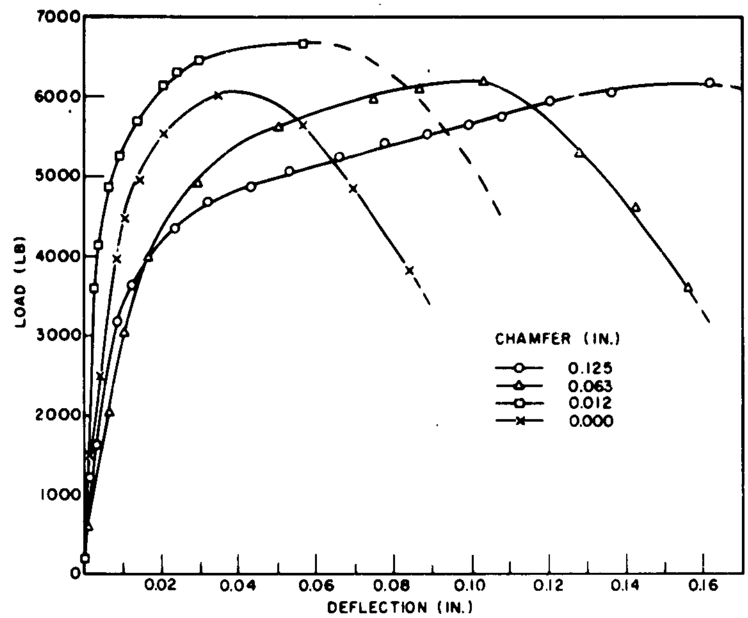
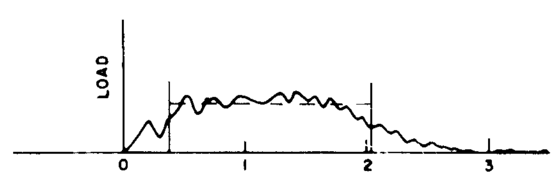


Fig. 8 - Load-deflection curves for SAE 1030 dowel specimens tested statically in shear

Dynamic tests were also made for all combinations of the two materials, two diametral clearances, and four chamfers. After each blow the distance between the two gage dowels attached to the test fixture was measured, the change in the measured value from its initial value represented the shear deformation caused by the plastic deformation. Load-time curves were obtained from the strain-gage load sensor from which such factors as stiffness, maximum load, and average load could be obtained. A representative curve of this type is shown in Fig. 9. An average load that is assumed to be effective during the plastic deformation is shown as the dashed line on this figure. The product of this load (force) and the plastic deformation that took place for this blow represents the energy required to produce this deformation.

Fig. 9 - Load-deflection curve for a dynamic test of a dowel specimen in shear



As an example of the type of response encountered during the impact tests, the average load (flow stress) and the plastic deformation are plotted in Fig. 0 as functions of the number of blows. The results shown are for SAE 1030 specimens with 0.0072 in. diametral clearance. Two sets of four curves are included, one curve for each of the four chamfers. The tests were terminated when the specimen fractured. From an examination of the load curves, it can be seen that the specimens tested with the larger chamfers required more blows to develop their maximum load and that this load was maintained for a greater number of blows. Again, it can be seen that all specimens developed about the same maximum load independent of chamfer size. The plastic deformation curves are all nearly linear and of the same slope, but the amount of deformation before fracture is greater for greater chamfer size.

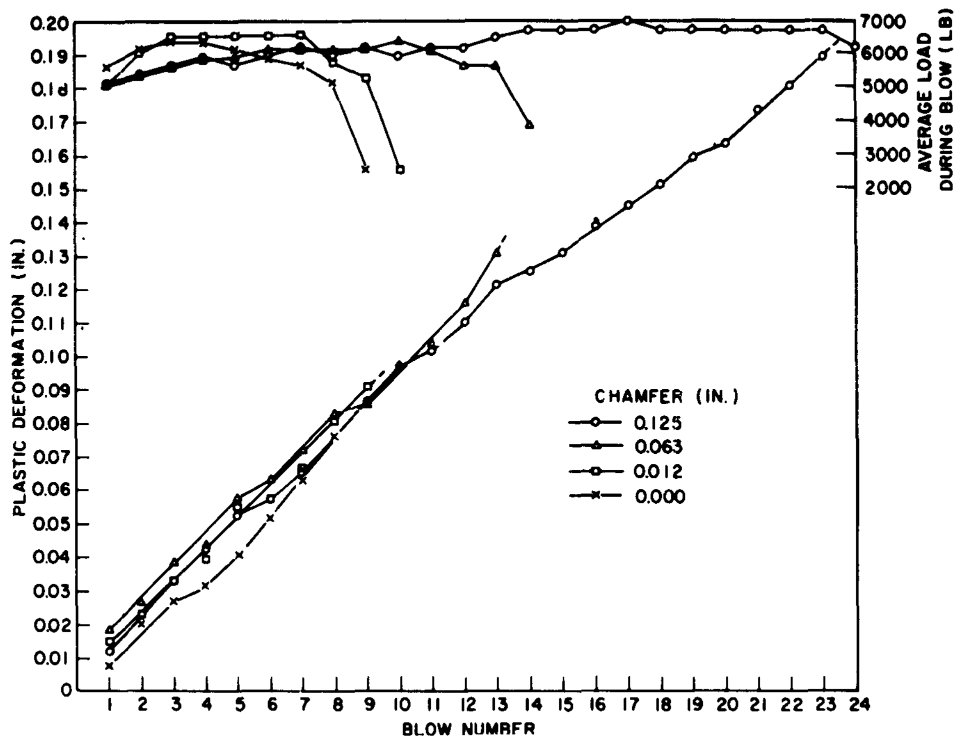


Fig. 10 - Average load and plastic deformation versus number of 4-in. hammer drops. SAE 1030 dowel specimens; clearance 0.0072 in.

The total amount of plastic energy absorbed prior to fracture versus chamfer size is shown in Fig. 11 for SAE 1020 and SAE 1030 steel for the two diametral clearances. The effect of clearance was not significant. In order to more easily compare the effect of the chamfer the curves of Fig. 11 are replotted in Fig. 12 as the ratio of the amount of plastic energy absorbed with chamfers to that without chamfers. A roughly linear relation is again observed.

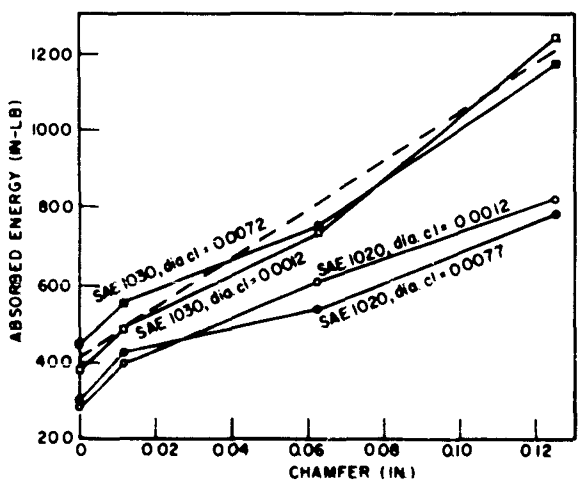


Fig. 11 - Absorbed energy versus chamfer size. The dotted lines represent a linear approximation and average for each specimen material.

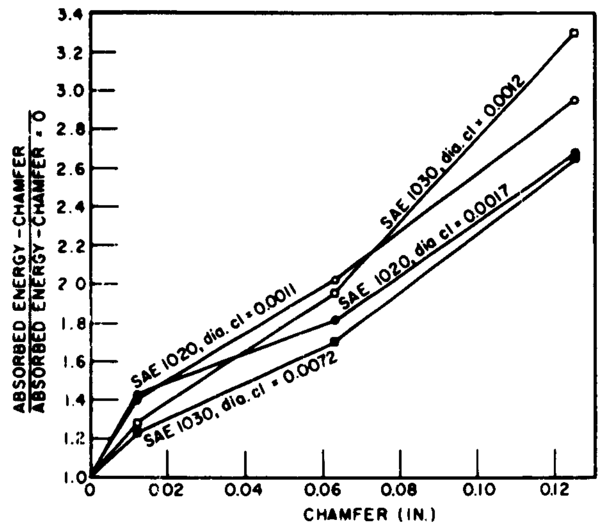


Fig. 12 - Ratio of energy absorbed with given chamfer to energy absorbed with no chamfer as a function of chamfer size

GENERALIZATION OF RESULTS

The relations shown in Figs. 5 and 11 can be represented by the equation

$$E_T = E_0 + E_C,$$

where E_T is the total energy absorbed by the bolt or dowel up to failure, E_0 is the energy absorbed when there is no chamfer, and E_C is the component of energy absorbed because of the chamfer.

If the chamfers are of fixed slope as shown on Fig. 13 (45-degree chamfers were used in the work described) and the total length of the chamfers, as projected along the bolt axis, is $L = L_1 + L_2$, then the bolt volume undergoing shear deformation as a result of the chamfer is $\pi D^2 L/4$. E_C should therefore be directly proportional to L. The value of E_0 is assumed constant. If the bolt diameter D should change, both E_C and E_0 should vary approximately as D^2. This, of course, assumes the bolt material remains constant. If one should wish to scale the above energy equation to obtain results for different bolt sizes, one can write

$$E_T = D^2(K_0 + K_C L),$$

where the values of K must be experimentally determined for each bolt material. From Fig. 5 for the 3/4-in. Navy Grade-2 bolts, this equation is determined to be

$$E_T = D^2(4450 + 48400\ L)\ \text{in.-lb.}$$

From the dashed line of Fig. 11 for the SAE 1030 steel dowels, but with the energy taken per single shear surface, this equation becomes

$$E_T = D^2(3280 - 50400\ L)\ \text{in.-lb;}$$

and for the SAE 1020 cold-rolled steel dowels, per single shear surface, this equation is

$$E_T = D^2(2560 - 31200\ L)\ \text{in.-lb.}$$

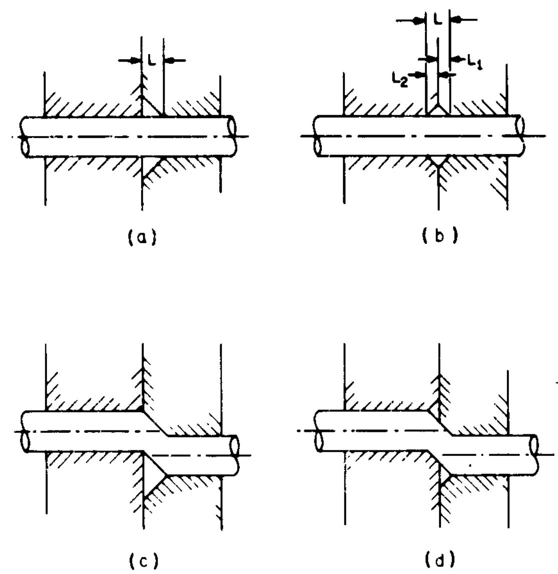


Fig. 13 - Chamfer arrangements. (a) single 45-degree chamfer of length L, (b) double 45-degree chamfer of total length L, and (c and d) bolt deformed by shear motions to correspond to chamfer shapes (a) and (b), respectively.

There are probably size effects and other approximations which are not accounted for in the above equation, so it should not be considered as giving more than an empirical estimate of the energy absorbed.

If one should wish to estimate the shear displacement that can take place for various size chamfers of the type illustrated in Fig. 13, good estimates will be obtained by merely adding the shear displacement permitted by the chamfer to the shear displacement the bolt would have for no chamfer, which in this case is

$$X_S = L + X_0.$$

For the illustrated chamfers, the chamfer angle is 45 degrees; hence the shear displacement permitted by the geometry of the chamfer is equal to L. X_0 is the shear displacement which would occur if no chamfer were present.

If one checks the above generalization for shear displacement with the information contained in Fig. 10 (where it is seen that X_0 = 0.075 in.), the following tabulation is obtained:

Chamfer Size (L):	0	0.012	0.063	0.125
Shear Displ. with Chamfer (X_S):	0.075	0.091	0.130	0.189
Calc. Chamfer Size (X_S - X_0):	0	0.016	0.055	0.114

The difference between the values of L and the corresponding value of X_S - X_0 indicates the degree of probable accuracy that might be expected. Of course the bolt material must be sufficiently ductile so as not to fracture prematurely in a brittle manner. It is probable that high-alloy steel bolts, heat treated for maximum tensile strength, will fail before full-shear displacement takes place if the chamfer size is large. The shear displacement subjects the bolt to considerable tensile stresses.

The average flow stress (load) preceding fracture appears to be relatively independent of the chamfer size. This is illustrated for a specific case by the upper set of curves in Fig. 10. This condition is required if the preceding energy and displacement relations are to be compatible.

CONCLUSIONS

For equipment having plane surfaces held together by bolts, chamfered bolt holes on the contacting surfaces permit greater absorption of shock energy by allowing greater shear deformations before fracture.

The amount of additional shear deformation is approximately equal to that which is permitted by the geometry of the chamfer. The average load, or flow stress, during the plastic deformation is relatively independent of chamfer size. Hence, the energy absorbed for shear motions of the surfaces is equal to a constant plus a factor which is proportional to chamfer size.

There is a significant but small decrease in stiffness of the joint because of the chamfer. However, the maximum load supported by a bolt in chamfered holes is as great as that supported by a bolt in an unchamfered hole.

REFERENCES

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Naval Research Laboratory. Report 5797. EFFECT OF CHAMFERED HOLES ON THE RESISTANCE OF BOLTS AND DOWELS TO SHOCK LOADS IN SHEAR, by I. Vigness, E. R. Seibert, and H. M. Forkois. 13 pp. and figs., July 9, 1962.

Under dynamic loading conditions associated with shock motions, hold-down bolts may be subjected to both shear and tensile stresses. If the bolt holes in the shear plane are chamfered so as to permit the bolt to deform in shear and bending along its length, the amount of energy absorbed by the bolt before fracture is increased when shear motions are encountered. The amount of energy absorbed when a 45-degree chamfer was employed was found to be a constant (the energy absorbed by a bolt in shear with no chamfer) plus a

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2. Bolts - Deformation
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